



25th ABCM International Congress of Mechanical Engineering
October 20-25, 2019, Uberlândia, MG, Brazil

COB-2019-2188

DEVELOPMENT OF A THERMAL-HYDRAULIC SIMULATION MODEL FOR ANGRA 2 AND 3 NUCLEAR POWER PLANTS

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Abstract. *In a world of digitalization of power generation and fourth industry revolution, the demand for computational tools that can give support to the continuous improvements of industrial plants is increasingly high. In such a context, a computational code for thermal-hydraulic simulations is being developed to serve as a predicting tool for Angra 2 and 3 nuclear power plants. The objective is to build a model capable of providing process data – temperature, pressure and mass flow – under steady state and transient regimes by means of zero and one dimensional approaches, i.e., at very low computational cost. This paper describes the first stage of development of this numerical code, showing the mathematical modeling of each component that compose the power cycle. The validation process and the results for some special cases are also presented.*

Keywords: *Numerical Simulation, Nuclear Energy, Thermal-Hydraulic Processes*

1. INTRODUCTION

In a time of digitalization of power industry – fourth industrial revolution – it is unquestionable that computational tools for process simulation are of the highest relevance in the pursuit for efficiency. As computing machines become more sophisticated, greater are the possibilities for engineers to build digital models of power plants and predict new scenarios. The demand for such predictions exists not only for new projects, but also for operating plants.

At the nuclear station of Angra dos Reis, both situations occur: Angra 2 and Angra 3 are twin plants, but the former has been operating since 2001 and the latter is under construction. For an existing plant, there are frequent design modifications, for improvements or adaptations, which present the necessity of understanding the system's behavior prior to such modifications, in the phase in which its impact on the system is evaluated. For a new project the demand for a simulation tool is evident. Besides assisting in the design process of the systems, it is used in the demonstration of accident scenarios to the licensing authority.

In this context, following such demand, a computational code for thermal-hydraulic simulations is being developed to serve as a predicting tool for Angra 2 and 3 nuclear power plants. The object is to develop a model capable of providing process data – temperature, pressure and mass flow – under steady state and transient regimes by means of zero and one dimensional approaches, i.e., at very low computational cost.

It is known that there are many commercial codes which perform the same task (Alobaid *et al.*, 2016), but the use of such programs has two drawbacks: one is the cost, which are often considerably high; and the other is that they are general codes which have to be customized to represent your system, which can be quite a complex task. Hence, the idea of building an *in-house* code is to have a perfectly adapted simulation tool developed specifically to Angra 2 and Angra 3 designs, which does not require a graphical user interface build up the system streams and has a specific model for each component.

The objective of this paper is to describe the computational code under development for thermal-hydraulic simulations of Angra 2 and 3, showing the mathematical modeling of each component, the validation process and the results for some special cases. In its current state, the code is capable of calculating the steady state process variables of the whole power cycle. The component models were configured using the Angra 2 parameters, i.e., heat transfer coefficients, areas, pipe length etc. To convert the code for an Angra 3 simulator it is only necessary to replace these parameters by the Angra 3 equivalents, since the main characteristics of the components are just the same (same type and number of heat exchangers, pumps, tanks etc.).

In the next section the system simulated by the code at the current stage of development is described. The following section presents the mathematical models employed for each component. Then the validation results and the simulation data provided for two scenarios – power uprate and heat exchanger plugging in one high pressure train – are presented.

2. DESCRIPTION OF ANGRA 2 AND 3 POWER CYCLE

Angra 2 and 3 are Pressurized Water Reactor (PWR) nuclear power plants with electric power generation capacities of 1,350 MW and 1,405 MW, respectively. In the PWRs, the nuclear reactor is cooled by a primary cycle, with subcooled water, which transfers heat to a secondary cycle through steam generators. The secondary cycle is responsible for the conversion of thermal power into mechanical and electrical power. The share of thermal power not converted into work is transferred to a tertiary (open) cycle.

The secondary cycle operates according to a Rankine power cycle with reheat and regeneration (fig. 1): steam is produced in the steam generators (SG), enters in the high pressure turbine (HPT) and flows to the moisture separator-reheater (MSR). Then the slightly superheated steam flows to the low pressure turbine (LPT), going through several expansion stages until reaching the main condensers (MC), where it condenses, rejecting heat to the refrigerating circuit (tertiary system, which is sea water). The condensate generated in the main condensers is pumped by the condensate pumps (CP) to the low pressure pre-heating trains (LPPH), in which the condensate absorbs heat supplied by the low pressure steam extractions. The feedwater tank (FWT) receives the main condensate as well as the condensate of the high pressure extractions. This tank supplies feedwater through the main feedwater pumps (FWP) to the high pressure preheaters (HPPH). The preheated feedwater is supplied to the steam generator, closing the cycle.

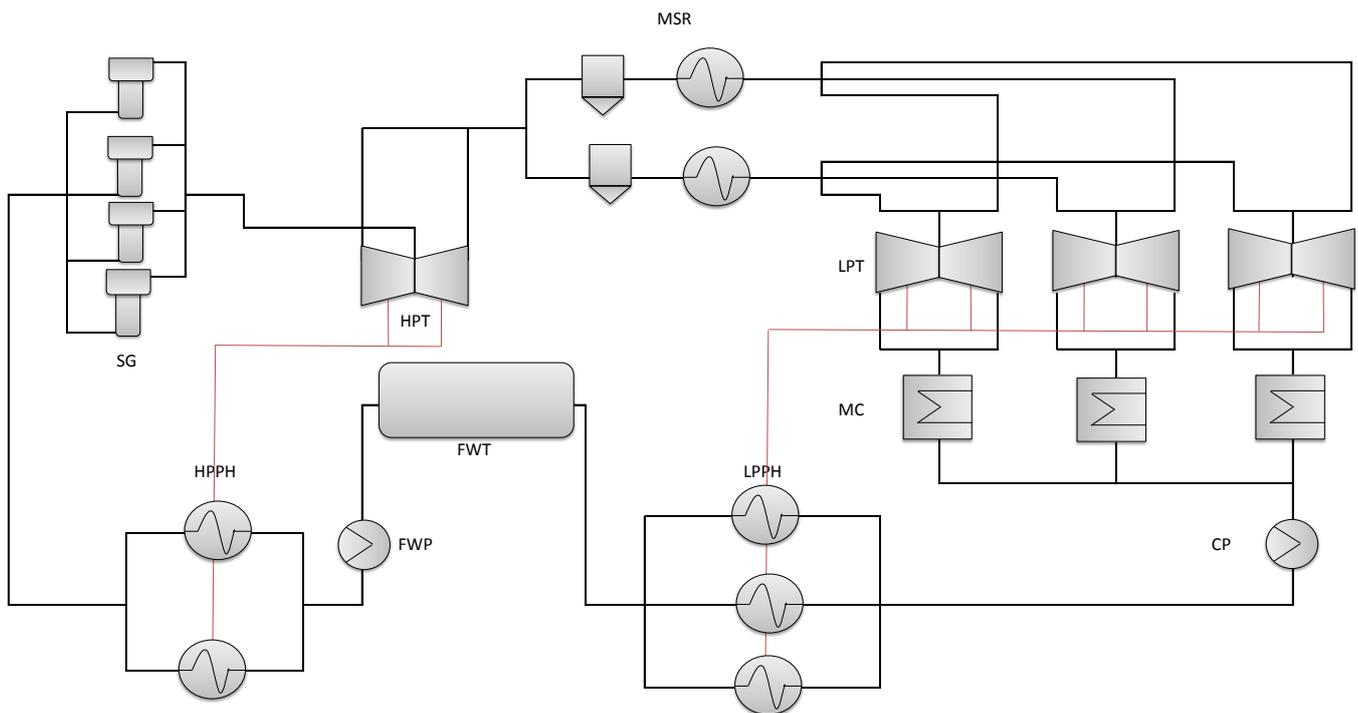


Figure 1. Power cycle (or secondary cycle) of Angra 2 and 3.

3. MATHEMATICAL MODELS

The computational code is composed by

- thermal balances in the heat exchangers with phase-change and without phase-change (coolers)
- thermal balances in the three-fluid heat exchangers (duplex)
- thermal balance for the feedwater tank
- thermal balance for the steam generator

- model for the steam expansions in the high pressure (HP) turbine
- stochastic models for the low pressure (LP) turbine
- stochastic model for the moisture separator-reheater

All these models are coupled to hydrodynamic balances, which are essentially the pressure losses along the lines. In the current stage of the model, a total steam mass flow enters as an input together with sea water temperature. Then the whole secondary cycle heat balance is calculated. One of the major challenges in the development of this model is the calculation of the extractions mass flow, which are free, i.e., they fluctuate freely according to the system's operating condition, following a non-linear coupling between pressure drop and thermal balance, through the mass flow of the extractions.

This issue was solved by means of an iterative procedure, in which the inlet pressure of the extraction steam is corrected with the turbine model. More details are provided in the sec. 3.7.

3.1 Coolers

Both the high pressure and low pressure pre-heaters are shell and tube type heat exchangers. The thermal model consists of the energy balance and the log mean temperature equations, i.e.,

$$\dot{Q} = [\dot{m}c_p(T_{\text{out}} - T_{\text{in}})]_{\text{shell}} \quad (1a)$$

$$\dot{Q} = [\dot{m}c_p(T_{\text{in}} - T_{\text{out}})]_{\text{tube}} \quad (1b)$$

$$\dot{Q} = UA\Delta T_{\text{ln}}F \quad (1c)$$

where \dot{m} is the mass flow, c_p is the specific heat at constant pressure, U is the overall heat transfer coefficient, A is the heat exchange area, ΔT_{ln} is calculated for the counterflow configuration and F is a correction factor (Shah and Sekulic, 2003).

The main challenge in solving this system of equation is to calculate the value of F , which changes according to the design and outlet temperatures. F is usually written as a function of P and R . While R is a constant, P depends on the outlet temperatures. According to Shah and Sekulic (2003), we have that

$$F = \frac{D}{1 - R \frac{\ln[(1 - RP)/(1 - P)]}{\ln[2 - P(1 + R - D)] - \ln[2 - P(1 + R + D)]}} \quad (2)$$

where $D = (1 + R^2)^{0.5}$

Thus it is necessary to use an iterative method to calculate the outlet temperatures and F all together. In the sequence of the calculating procedure, an initial value for P is estimated. With the energy balance (eqs. 1) and the one for F (eq. 2) it is possible to get new outlet temperatures and a new value for P , which is the value used at the next iteration.

3.2 Duplex coolers

At the beginning of the low pressure train (LPPH), there is a three fluid heat exchanger whose analysis had to be different from the previous ones. The duplex heat exchangers have only one shell for two tube sheets and the idea used for the coolers are not adequate for this component. Therefore, to estimate the outlet temperatures, the ϵ - NTU concept is taken into account. According to Shah and Sekulic (2003), we have that $NTU = UA/C_{\text{min}}$.

$$\epsilon = \frac{2}{1 + C + (1 + C^2)^{0.5} \coth(\Gamma/2)} \quad (3)$$

where $\Gamma = NTU(1 + C^2)^{0.5}$ and $C = C_{\text{min}}/C_{\text{max}}$ and

$$\epsilon = \frac{(T_{\text{out}})_{\text{tube}} - (T_{\text{in}})_{\text{shell}}}{(T_{\text{in}})_{\text{tube}} - (T_{\text{out}})_{\text{shell}}} \quad (4)$$

Therefore, the outlet temperatures can be calculated.

3.3 Main condensers (MC) and pre-heaters (LPPH/HPPH)

Although these components also present shell and tube design, the approach was different because the shell flow is condensing. In the MC, as expected, the main steam flow is condensed, while in both the high pressure and low pressure lines, the pre-heaters are used for condensing the turbine's extractions. A reasonable approach was to consider that the steam side only suffers a phase change, thus the temperature is constant. Therefore, the only unknown outlet temperature can be calculated through an energy balance, i.e.,

$$\dot{Q} = [\dot{m}c_p(T_{out} - T_{in})]_{tube} \quad (5a)$$

$$\dot{Q} = [\dot{m}(h_{in} - h_{out})]_{shell} \quad (5b)$$

This solves the model, since in this problem the \dot{m} from the turbines is not an input anymore. The flow can actually be calculated together with the tube outlet temperature and \dot{Q} . The turbine analysis is detailed at the subsection 3.6

3.4 Steam Generators (SG)

The analysis was developed using a different approach than the ones employed for the turbine extractions heat exchangers. The main difficulty regarding the Steam Generator is in the subcooled and phase change regions, as the equations that govern this two phenomena are distinct. Moreover, as we can see in fig. 2, the subcooled region is divided into a parallel flow part and a counterflow part, which also needed to be handled differently.

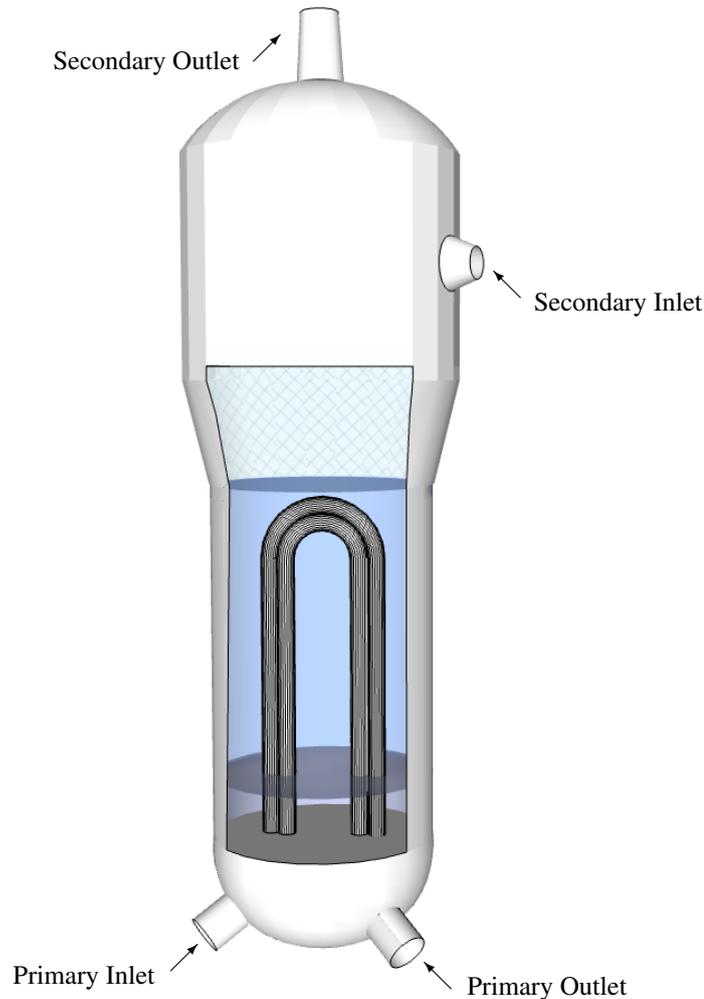


Figure 2. Detail of subcooled region inside the steam generators.

The following equations were used for the subcooled region, according to Oliveira *et al.* (2018).

$$\left(\frac{\dot{m}_p c_{pp}}{\dot{m}_s c_{ps}} - 1\right) \frac{dT_p}{dA} + U\mathbf{X}' \left[T_{si} - \left(\frac{\dot{m}_p c_{pp}}{\dot{m}_s c_{ps}} + 1\right) T_p + \frac{\dot{m}_p c_{pp}}{\dot{m}_s c_{ps}} T_{pi} \right] = 0 \quad (6a)$$

$$\left(\frac{\dot{m}_p c_{pp}}{\dot{m}_s c_{ps}} - 1\right) \frac{dT_p}{dA} + U\mathbf{X}'' \left[T_{si} + \left(\frac{\dot{m}_p c_{pp}}{\dot{m}_s c_{ps}} - 1\right) T_p - \frac{\dot{m}_p c_{pp}}{\dot{m}_s c_{ps}} T_{po} \right] = 0 \quad (6b)$$

applied for the parallel flow (eq. 6a) and counterflow (eq. 6b), where:

$$X' = \frac{1}{m_p c_{pp}} + \frac{1}{m_s c_{ps}} \quad (7a)$$

$$X'' = \frac{1}{m_p c_{pp}} - \frac{1}{m_s c_{ps}} \quad (7b)$$

and T is temperature, A is the heat transfer area, \dot{m} is the mass flow and c_p is the specific heat at constant pressure. The subscripts p and s refer to the primary and secondary systems, while i and o refer to inlet and outlet temperatures, respectively.

3.5 Head loss

The hydraulic simulation was developed through a careful survey of the pipelines in which all of local and dynamic head losses were taken into consideration. The models described in Crane Co. (2013) were used for calculating the local hydraulic resistances for each component. The total mass flow in the system is given, in order to compute the mass flow for each train, two equations were elaborated. Firstly, to measure the total hydraulic resistance K_t in each train, considering area changes during the pipeline, we have:

$$\frac{K_t}{A_m^2} = \frac{K_1}{A_1^2} + \frac{K_2}{A_2^2} \quad (8)$$

where K_1 is the resistance with area A_1 and K_2 is the resistance for area A_2 . The area A_m is the average area, which considers the pipe length in A_1 and A_2 .

Once K_t is obtained, the following equation was employed for trains in parallel arrangement

$$\frac{A_t}{\sqrt{K_{eq}}} = \sum_{i=1}^n \frac{A_i}{\sqrt{K_i}} \quad (9)$$

for a system with n trains with index i . With K_{eq} it is possible to measure the total head loss Δp . By knowing that it has to be the same in each train, we can finally evaluate how the total mass flow is divided.

3.6 Steam Turbine

In the steam turbine, the model calculates the outlet pressure for each expansion stage. The HP turbine has two expansion stages and the LP turbine has six expansion stages. Therefore, eight outlet pressures are calculated.

These pressures are calculated by employing the ellipse model, using the Stodola coefficients (Cooke, november 1983; Dettori *et al.*, 2017). It consists on the assumption that, for each expansion stage,

$$\frac{W^2 T}{p} = \text{constant} \quad (10)$$

where W is the inlet mass flow, T is the absolute temperature and p is the total pressure. Dettori *et al.* (2017) extend this relation to an expression relating the inlet and outlet pressures of a particular expansion stage, represented by

$$W = K \sqrt{\frac{p_{in}^2 - p_{out}^2}{T}} \quad (11)$$

in which the Stodola coefficient K determines the proportionality between both sides. There is one coefficient K_i associated to each expansion stage i , and they are obtained by the operating data of the turbine. Figure 3 shows the curves corresponding to the two stages of the HP turbine.

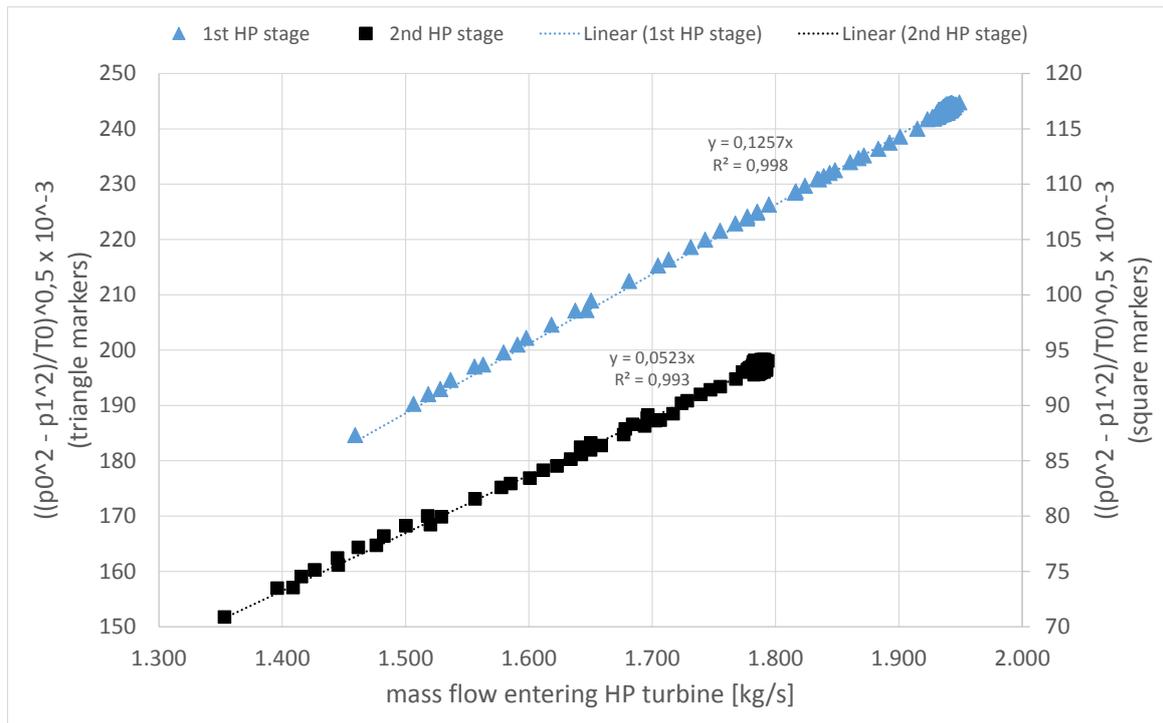


Figure 3. Determination of the Stodola coefficient of the HP Turbine.

3.7 Prediction of steam extractions

The secondary cycle has six steam extractions, designated A1, A2 ... A6, in growing order of pressure (A1 is the extraction with smallest pressure). Except for the A5, which is taken from the MSR, all extractions are taken from the turbine, with A6 coming from the HP turbine and A1 ... A4 coming from the LP turbine.

The mass flow of the extractions are not controlled, i.e., they fluctuate according to the operating conditions the turbine and in the pre-heaters. For example, if the feedwater mass flow increases, the pressure of extraction side (shell) of the pre-heaters decreases, resulting in a larger extraction mass flow. But this mass flow must satisfy the hydrodynamic restrictions, i.e., pressure loss, which has a quadratic relation with mass flow. This process also occurs with the last expansion stage, the one which flows to the main condensers.

This coupling is solved by an iterative procedure in which an inlet pressure of the extraction p_{ext} is guessed for the first thermal calculation in the pre-heater. The calculation flux continues and in the turbine, where outlet pressures are calculated, the inlet pressure p_{ext} is corrected. The thermal calculation in the pre-heater is performed again with the new pressure p_{ext} and the process goes on until convergence.

4. THE COMPUTATIONAL CODE

The model is implemented in VBA programming language, based on a structure of MS EXCEL workbooks. The structure is presented in fig. 4 by an UML diagram.

The code has an interface in EXCEL to assist the user in during the problem setup and the visualization of the results. Figures 5 and 6 show the main interfaces for results visualization.

5. RESULTS

Two cases were simulated using this model:

- normal operation based on the thermal heat balance of September 12th, 2018
- power uprate, based on a calculated thermal heat balance

Tables 1 and 2 show the results for the main variables of the secondary circuit heat balance. The values of the calculated electrical and thermal powers, pressures (p), mass flows (W) and temperatures (T) are presented and the

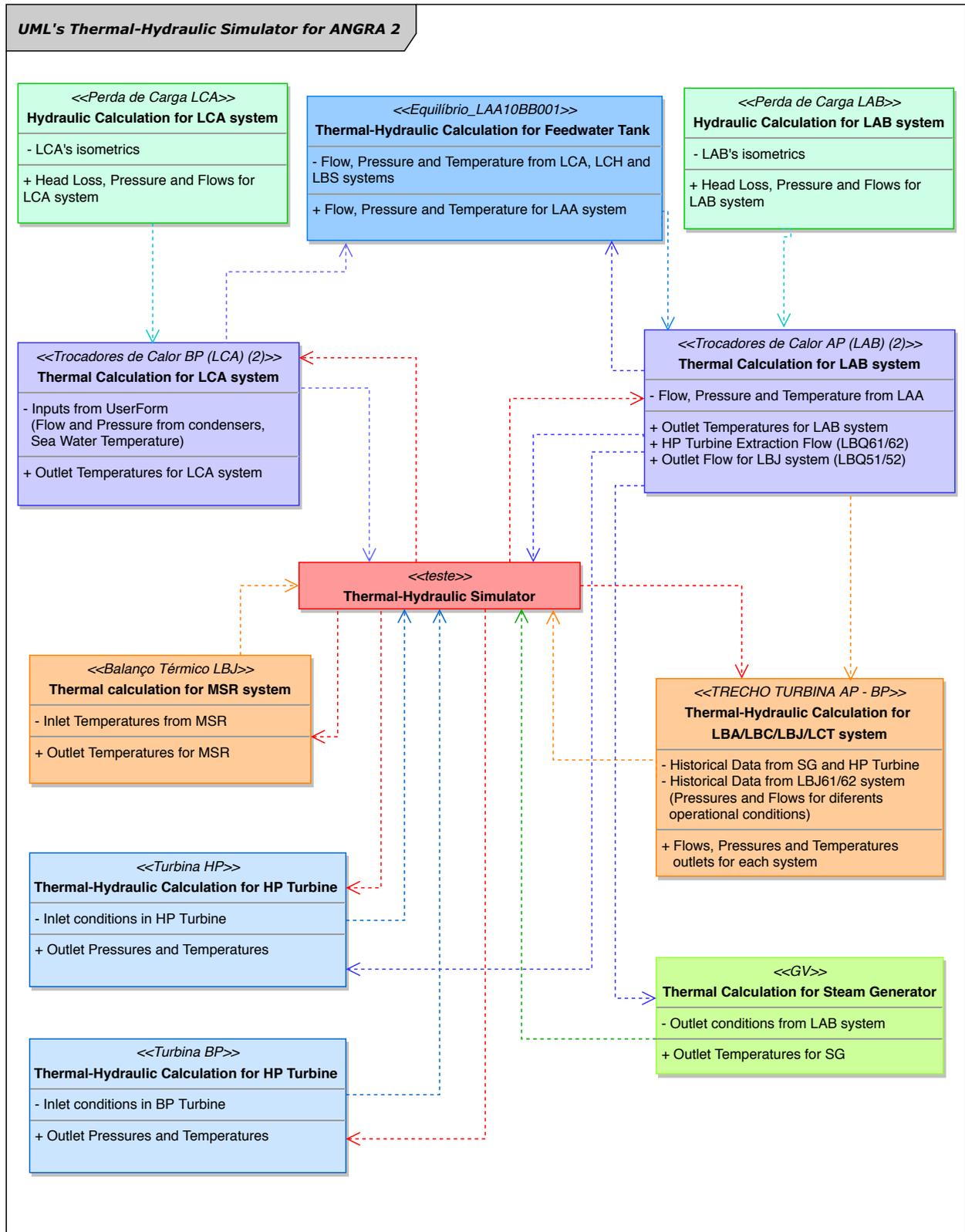


Figure 4. UML diagram of the computational code.

associated relative error, with respect to the reference thermal heat balance, is shown aside each value.

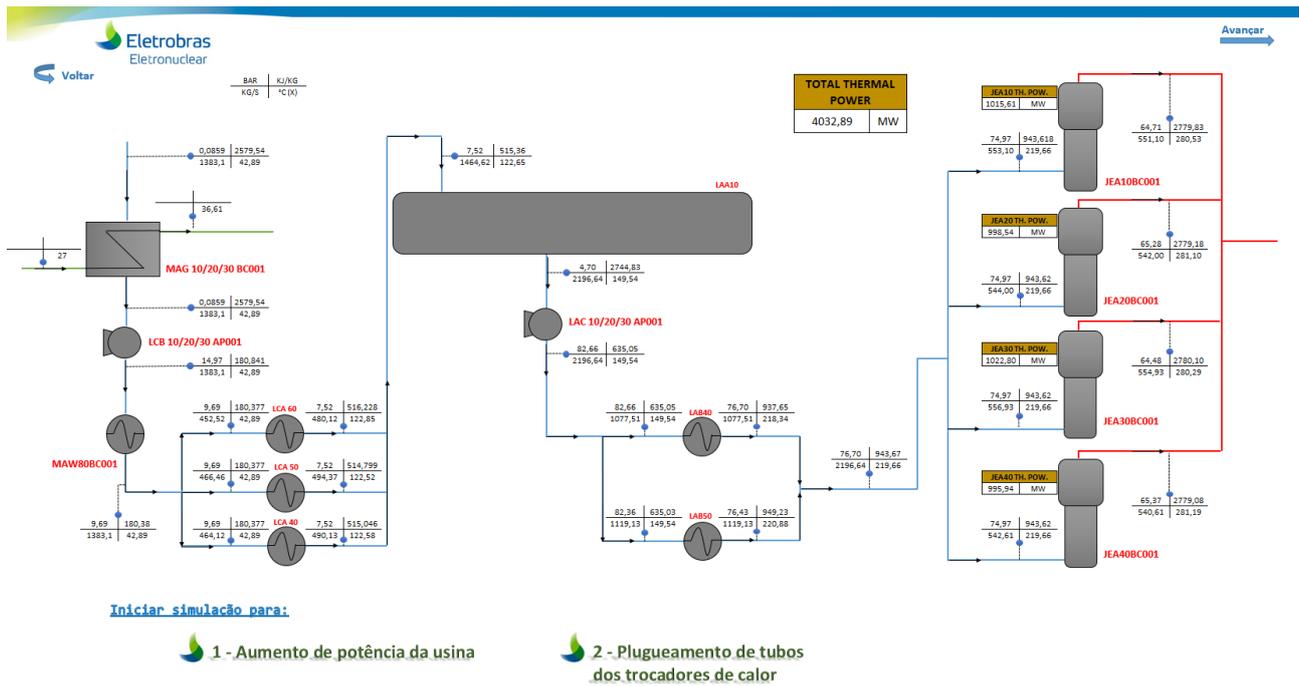


Figure 5. Graphical user interface of the pre-heating trains.

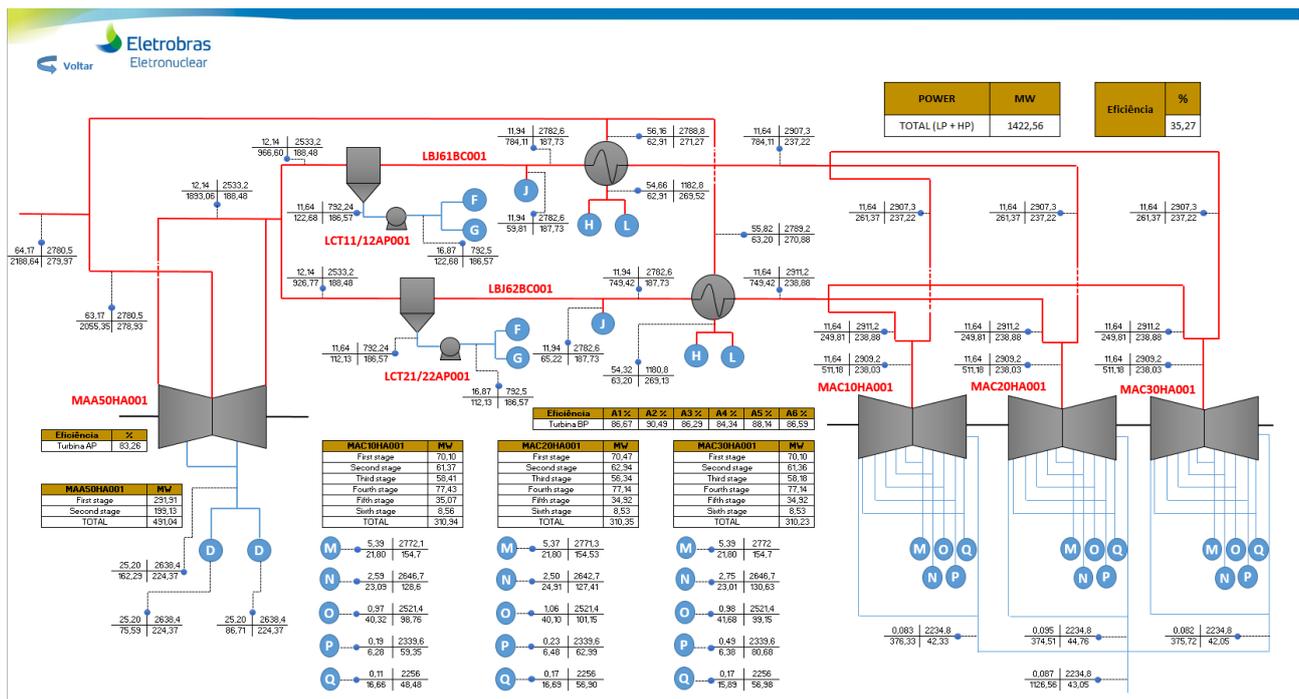


Figure 6. Graphical user interface of the MSR and turbines.

A global error E was calculated from the main process variables shown in tables 1 and 2, using the expression

$$E = \sqrt{\sum_{i=1}^{116} \left[\frac{(V_{calc})_i - (V_{ref})_i}{(V_{ref})_i} \right]^2} \quad (12)$$

where $(V_{calc})_i$ is the calculated value of the i^{th} variable and $(V_{ref})_i$ is the corresponding reference variable. It's some sort of norm 2 of the relative error vector. For the normal operation case, a global error $E = 26.14\%$ was obtained. The mean error was $\bar{\epsilon} = 2.17\%$.

For the power uprate case, the global error was $E = 19.12\%$, with mean relative error $\bar{\epsilon} = 1.81\%$.

Table 1. Simulation of normal operation in Angra 2. Results are compared against the heat balance of September 12th, 2018.

	power [MW]	%	p [bar]	%	W [kg/s]	%	T [°C]	%
thermal power	3837,50	1,27 %						
electrical power	1391,04	0,26 %						
condenser outlet			0,07	21,76 %	1326,89	0,00 %	38,76	7,19 %
CP outlet			15,42	1,51 %	1326,89	0,00 %	38,76	0,56 %
LPPH trains inlet			10,21	1,79 %	1326,89	0,58 %	38,76	1,12 %
LPPH trains outlet			8,21	2,88 %	1397,90	1,70 %	122,99	0,41 %
FW tank outlet			4,70	0,00 %	2074,54	0,17 %	149,53	1,10 %
HPPH trains inlet			83,88	0,56 %	2074,54	0,17 %	149,53	1,88 %
HPPH trains outlet			77,91	1,02 %	2074,54	0,17 %	216,59	0,89 %
extraction A6			23,63	4,25 %	143,98	3,34 %	220,97	1,03 %
extraction A5			10,85	7,97 %	112,68	4,22 %	183,48	1,97 %
SG inlet			76,36	0,99 %	2074,54	0,00 %	216,59	0,89 %
SG outlet			65,06	3,53 %	2074,54	0,00 %	280,88	0,86 %

Table 2. Simulation of normal operation in Angra 2. Results are compared against the heat balance of September 12th, 2018.

	power [MW]	%	p [bar]	%	W [kg/s]	%	T [°C]	%
thermal power	4032,89	1,18 %						
electrical power	1422,56	1,21 %						
condenser outlet			0,09	0,00 %	1383,10	0,00 %	42,89	0,00 %
CP outlet			14,97	–	1383,10	0,00 %	42,89	0,49 %
LPPH trains inlet			9,69	1,62 %	1383,10	1,09 %	42,89	0,49 %
LPPH trains outlet			7,52	0,92 %	1464,62	0,10 %	122,85	0,69 %
FW tank outlet			4,70	5,81 %	2196,64	0,02 %	149,54	1,36 %
HPPH trains inlet			82,66	9,48 %	2196,64	0,02 %	149,54	2,26 %
HPPH trains outlet			76,70	6,14 %	2196,64	0,02 %	219,66	0,47 %
extraction A6			25,20	0,64 %	162,30	2,76 %	224,37	0,15 %
extraction A5			11,94	1,08 %	125,03	11,34 %	187,73	0,27 %
SG inlet			74,97	3,75 %	2196,64	0,02 %	219,66	0,47 %
SG outlet			64,96	5,84 %	2196,64	0,51 %	280,7775	1,43 %

It must be mentioned that the errors are highly sensitive to the difference in the condenser pressures.

6. CONCLUSIONS

The first results obtained in this project of developing a numerical code for thermal-hydraulic simulation for Angra 2 and 3 were presented in this paper.

From the data shown in tables 1 and 2, it can be concluded that the model is already capable of providing thermal-hydraulic process data for Angra 2 and 3 with fairly low uncertainties. It can be seen that most of the calculated process data shown in the tables had a relative error smaller than 2%, including those for the power uprate case. The most sensitive part of the model are the low pressures at the end of the steam expansion, exhibiting the worst errors.

In addition, it is worth to mention that the code has been implemented in MS VBA. Therefore it runs in any machine with a MS Excel installed. The next steps of this project are implementing the calculations in the LP turbine (currently the model uses a stochastic approach for solving it) and transient simulation, evolving from the 0-D to the 1-D models.

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